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PRELIMINARY ASSESSMENT OF LIGHTER DAMPING AUGMENTATION IN HEAVE FOR REDUCTION OF RELATIVE MOTION

Sidney Berger

Control Systems Research, Incorporated

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The concept of augmenting the inherent damping of a lighter hull to suppress its heave response in waves, and thereby to reduce relative motion between a cargo ship and lighter is the subject of this investigation. The scope of the effort was a minimum one to rapidly assess the promise of the concept. A primarily empirical approach was taken. Sufficient analysis was included to interpret the test results and to consider the implications of the results in full scale.

Model tests were performed in a model basin with wave making facility. The model consisted of a one-sixth scale hull for a 45 ft. long patrol boat with its damping appendage slats, one or two high, longitudinally aligned on either side. Model mounting arrangements on the stationary carriage of the towing tank locked out all motions other than heave. Tests covered the model at two displacement conditions of bare hull, with single row damping, with double row damping. Wave lengths were in the range of 6 to 16 ft. Wave heights were between 2.6 and 5.0 inches, with the attempt made to obtain the steepest possible wave without waveform distortion.

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I-C

Preface

The investigation reported herein was performed by Control Systems Research, Inc. under Contract No. N00025-72-C-0044. The contractor effort was directed by Sidney Berger. Monitoring of the work for Naval Facilities Engineering Command was the responsibility of Milon Essoglou. The stimulating manner in which his function was carried out is fully appreciated by the contractor. Acknowledgement is also made of the workmanlike fulfillment of a subcontract by Hydronautics, Inc. for the provision of model basin facilities.

This contract, in addition to meeting its objectives, has reinforced the admonition of Prof. B.V. Korvin - Kroukovsky contained in his paper (cited in the text of this report):

> "The ship response to regular waves can be measured in a towing tank without any reference to theory. However, experimental data do not bring out readily the various physical properties of a ship. It is necessary to develop a reasonably comprehensive theory in order to gain an understanding of the effects of these properties on the motions of a ship. Such a theory also would provide a basis for judging the experimental data. There are many pitfalls in towing-tank testing of ship models in waves which can be avoided when knowing what to expect on theoretical grounds."

In the case at hand, the general form of heave response curves is widely appreciated. Nevertheless, there appears to be more speculation than precision in the explanation of the phenomena contributing to the heave response. The contractor believes that this investigation might heighten interest in the subject, leading to fuller understanding of heave response and ultimately to means of controlling the motions of lighterage in a practical way.

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SUMMARY AND CONCLUSIONS

1.1 The concept of augmenting the inherent damping of a lighter hull to suppress its heave response in waves, and thereby to reduce relative motion between a cargo ship and lighter is the subject of this investigation. The scope of the effort was a minimum one to rapidly assess the promise of the concept. A primarily empirical approach was taken. Sufficient analysis was included to interpret the test results and to consider the implications of the results in full scale.

1.2 Model tests were performed in a model basin with wave making facility. The model consisted of a one-sixth scale hull for a 45 ft. long patrol boat with its damping appendage slats, one or two high, longitudinally aligned on either side. Model mounting arrangements on the stationary carriage of the towing tank locked out all motions other than heave. Test covered the bare hull model, then with single row damping added, and with double row damping, all at two displacement conditions. Wave lengths were in the range of 6 to 16 ft. Wave heights were between 2.6 and 5.0 inches, with the attempt made to obtain the steepest possible wave with waveform distortion.

1.3 The test results show that the non-dimensional response, (model heave amplitude: wave amplitude,) decreases with wave frequency, from about 0.7 to 0.2. The highest response was for the higher values of wave length: ship length ratio. No striking differences between the heave response amplitude operator curves for the damped and bare hull models are apparent by superficial data inspection.

1.4 However, in the case of the bare hull for which the widest range of wave length points were taken, the response curve appears to have bottomed out and taken an upward trend. This would conform to the pattern of heave response curves which model tests generally show. In particular, the analysis at NCEL shows a heave response quite close to the measurements and a decided peaking beyond the depression in the curve at the region of wavelength equal to model length.

1.5 Declining amplitude tests were also performed. These results showed that the inherent damping of the bare hull had been increased about six-fold by the addition of the double-row slat arrangement. Concurrently the model with damping appendage has a reduced natural frequency (from 6.0 to 4.8 radians/sec.) obviously due to the large added mass associated with the damper.

1.6 Using the experimentally measured natural frequencies, the response was plotted in terms of tuning factor. The bare hull shows the same tendency and appears headed for a peak just beyond ω_e/ω_n of one. However, the damped model shows test points just beyond the one position and the response curve is at the lowest level.

1.7 Using the case of an LCU as a representative lighter in logistical cargo movement it may be seen that the full scale situation centers interest on the region of λ:L less than 0.6. For a six-foot wave the average length is 78.8 ft. and wave lengths even shorter will have greater frequency of occurrence. This means that the peak on the response curve should fall roughly coincident with an unfavorable part of the wave spectrum.

1.8 Conclusion. On the basis of work performed it is clear that the peaking in the heave response amplitude operator includes a manifestation of resonance and that damping will influence response in this region. In long waves, i.e. greater than λ: L of 1.3, there is no favorable result from damping and only wave length effects govern the motion. The damped model has a test at about where peaking in heave response should be evident but its amplitude response ratio remains at the lowest level. Furthermore, this is the region where full scale response effects on an LCU type will be most critical. It is therefore concluded that the concept of damping augmentation to suppress relative motion is a promising approach.

1.9 Recommendation. At this point the case to support the conclusion could be further reinforced as a prequisite to any substantial commitment of research and development resources. It is recommended that two specific tasks be undertaken:

- The existence of the computerized analytical model of ship motions at NCEL provides a ready means of investigating the form of the heave response amplitude operator. Various systematic changes in model properties (such as displacement, added mass coefficient, and damping coefficient) can be made to observe their influence on the peak of the response curve. Analytical understanding of the phenomena should be pursued concurrently with test.

- Further tests are warranted to expand the response curves into the region of higher exciting frequencies, i.e. shorter waves. This

means lower wave amplitudes and more careful test technique (for example, more tank stilling between runs). The models should be designed for experimental investigation of the peak in the response curve by systematic variation in natural frequency and damping coefficient. Lessons learned in the first runs should be incorporated in the models. The slats should be non-buoyant, and should be at a greater depth to avoid any wave orbital motion acting on the slats to increase heave excitation, as may have occurred in the first tests. Test planning should include continuous interchange of thinking with the above analysis task.

Section 2

BACKGROUND AND OBJECTIVES

Conspicuous among the unsolved problems of cargo transfers in the military environment is the landing of a cargo draft on a lighter or other ship-to-shore craft from an ocean-going cargo liner, either merchant or naval auxiliary. In unprotected waters the sea surface disturbances drive various motions of the ship and lighter. The two vessels respond quite differently to the driving forces and relative motion of the lighter with respect to the ship can be substantial. As a consequence of this relative motion the rate of cargo transfer to shore can be degraded--especially as sea conditions worsen. Under severe conditions the cargo rate can fall to point where the risk of a major accident does not justify the deliveries that can be achieved and operations are totally suspended.

2.1 Related Background

A number of attempts to alleviate the relative motion problem are general knowledge. They can be characterized as overly ambitious and encumbered by non-essential complexity. One approach is a feedback servo system in which the motion of the lighter is sensed and a compensatory motion is applied to the load line. The Patent Office contains a disclosure of a device for hoisting a lighter out of the water from the ship. This arrangement obviously simplifies the cargo transfer but replaces the one problem with the equally difficult task of engaging and hoisting the lighter.

These representative examples of prior approaches to the relative motion problem involve significant modifications to the cargo ship which, of course, is not necessarily dedicated to moving military cargoes to unimproved sites if it is a merchant liner. There are the additional questions of subsidy payments and in probable utilization of any specific ship. In short, the prior art does not appear to include any devices, or even approaches, which encourage the hope that their effective contribution to offshore ship unloading would offset their disproportionately high costs.

2.2 The Concept of Damping Lighter Heave Motions

The designs of current and contemplated landing craft and cargo lighters are essentially a compromise between their transit and their beaching characteristics. Admittedly it is difficult, within cost constraints, to achieve high performance in the several modes of operation that a cargo lighter experiences. Nevertheless, there is no evidence that any consideration at all is given, during lighterage design, to providing as stable a platform as possible for landing the cargo coming off the ocean-going ship. This, in effect, is a third phase in the full cycle of lighterage operation.

The possibility is thus raised that a means can be found to incorporate motion damping into the craft design. If the device were to be passive it would have a decided advantage over any active control system. There is no clear requirement that such a device would need to aim for complete cancellation of the relative motion between lighter and ship. Any significant suppression of lighter heave motions would expedite cargo transfers by easing the burden on crane operators to manually provide the compensation to the load line.

As a first step in evolution of a passive motion damper the use of flat plate drag was proposed. An array of slats might conceivably be disposed below the hull of an LCU-type craft as shown in Figure 1. The slats could be mounted on struts

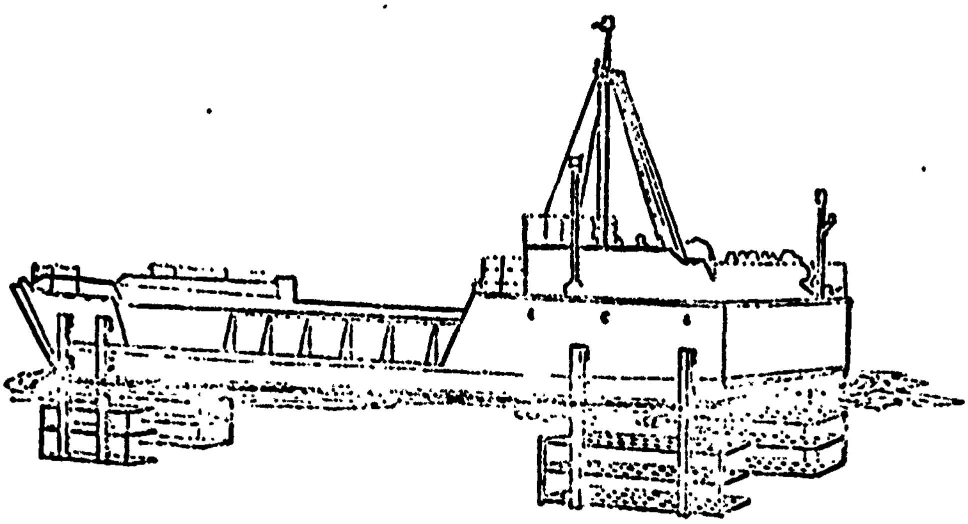


Figure 1 - Potential Approach To Damping Augmentation

supported off of fittings affixed to the hull of the craft. Ultimately, some means would be needed to retract the drag surfaces during transit to shore and beaching.

2.3 Objectives of Present Task

The work reported herein has been performed to assess the potential merit of the concept of passive damping as a means of suppressing relative motion between lighter and ship. The specific objective was taken to be a series of tests in a hydrodynamic wave generating basin. Heave motions only were to be covered by the tests. The utmost simplicity in model construction was to be observed. In short, the work was intended as an expedited, essentially empirical first attempt to damp the heave motions of lighters in waves.

Section 3

TEST NARRATIVE

While the subject of ship motion response to the forcing effect of waves is anything but simple, the limited objectives (and resources available) in this project dictated a bare minimum of theoretical preparations for testing. Nevertheless, the models were configured to illustrate the dominant effects. Sufficient analyses were performed to size the damping device and set the test parameters.

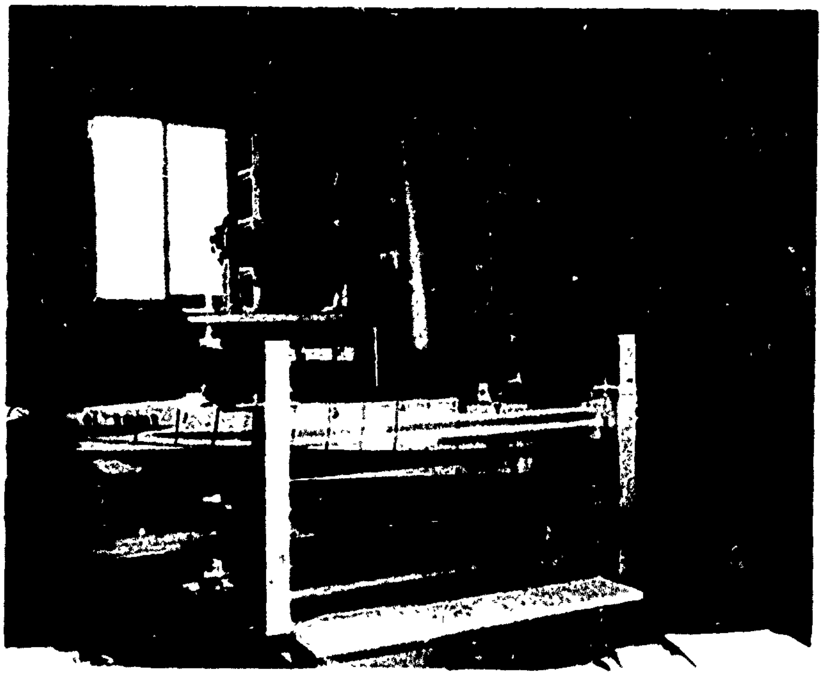
3.1 Model Description

In the expectation that the damping appendage of the model would dominate the inherent damping of the hull, no special requirements were placed on the hull for the purpose of these tests. A one-sixth scale model of 45-foot Army patrol boat was available and was used. Frames were designed to facilitate the mounting and demounting of the damping slats. The slats are located longitudinally along either side either singly or a two-row stack. A view of the model with maximum damping installed is shown in Figure 2.

Several characteristics of the hull are relevant to the damping tests. In order to predict the undamped natural frequency, the heave spring rate in terms of tons per inch immersion is required. For full scale, the hull data used in preliminary calculations is taken from Figure 3. Using a tpi value of 0.965 from the curve (corresponding to normal displacement of 15.5 tons) and an initial estimate of added mass equal to displacement mass, the natural period in heave for the full size craft is found to be $\tau = 1.80$ seconds. For the one-sixth scale model the period $\tau = 0.74$ seconds (corresponding to a natural frequency of 1.35 hertz).

The structure in general and the damper support frame are obviously over-designed. The approach in assembling the model was to assure sufficient ruggedness with only a hasty qualitative estimate of loads during handling and testing.

a - With Damping Appendage Installed on Weighing Platform



b - On Carriage In Waves

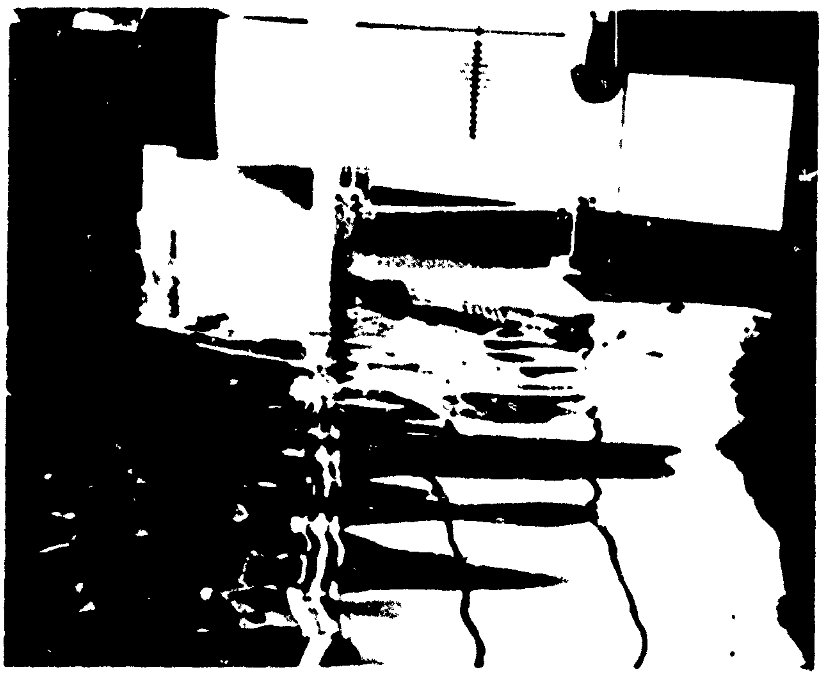


Figure 2 - Model General Arrangement

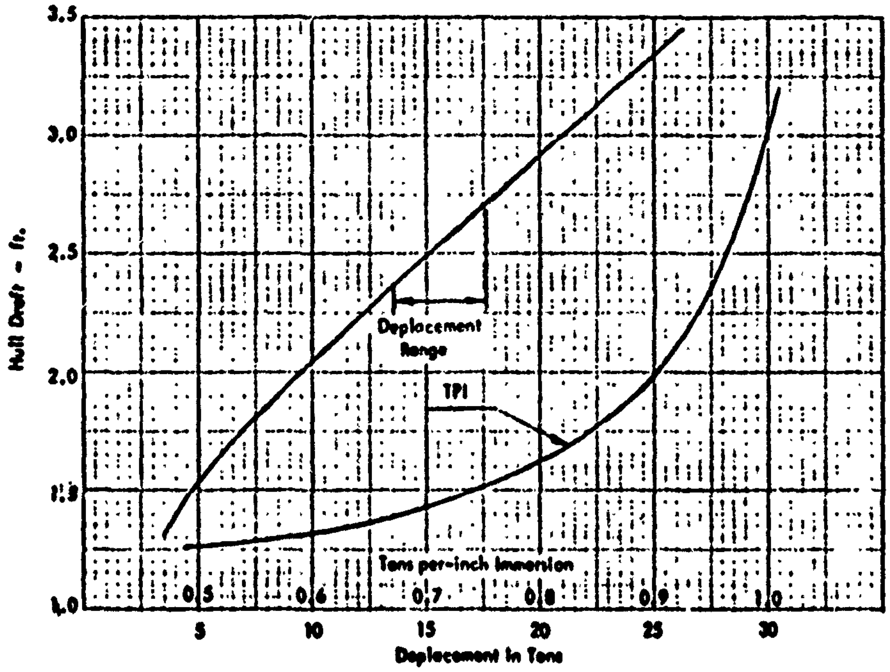


Figure 3 - Hull Data for Frequency Prediction

One rather unfavorable consequence was that the selection of three-quarter inch thick plywood for the damping slats amounts to 1.25 cu. ft. and necessitated 33 pounds of additional ballast to the model with damper, for a given displacement condition. Thus, this extra mass, together with the much greater added mass of the model with damper, complicated the comparison of the performance with and without dampers.

3.2 Preliminary Estimate of Damping

A simplified analysis was performed to obtain a preliminary estimate of the amount of damping to be incorporated in the model. Despite the frequency dependence of hydrodynamic damping, only a single point value was computed. Consider the case at the predicted resonance frequency (8.47 radians/sec.), and an assumed single amplitude of 0.25 ft. For harmonic oscillations the peak velocity is:

$$\begin{aligned} y &= \omega y_o \cos \omega t = 8.47 \times 0.25 \\ &= 2.11 \text{ fps.} \end{aligned}$$

Designating the coefficient of the velocity term in the equation of motion for a single degree of freedom elastic system as "C", the critical value of damping (the point at which the motion just loses its vibratory character) is:

$$\begin{aligned} C_{cr} &= 2\sqrt{\frac{KW}{g}} = 2\sqrt{\frac{712 \times 320}{32.2}} \\ &= 169.4 \text{ lbs./fps} \end{aligned}$$

The peak damping force that would be developed is a function of the highest velocity reached in an excursion. For the particular case used in the estimate the force is found to be 359 pounds, from which the equivalent flat plate drag can be found. In this case the drag force is

$$D_p = C_p A\left(\frac{\rho V^2}{2}\right)$$

For a rectangular plate the coefficient of parasite drag (C_p) varies with aspect ratio between approximately 1.1 and 2.0. A value of 1.25 is appropriate for the dimensions seen in the assembled model--from which it can be determined that 66.5 sq. ft. of plate area would result in critical damping. Dimensions of the slats are 7 ft.-6 in. long by 8.0 inches wide for a total of 20 sq. ft. for the four elements.

The final step in the damping estimate concerns the effectiveness of the surfaces. The width of a single slat could have been increased and while the drag coefficient would have decreased there would be a net gain with greater surface. However, stacking with inadequate separation is not efficient. When the separation distance is less than the least dimension of the plates, the second plate can reduce the total drag of the assembly, apparently due to the reduction of eddy formation in the wake. For the separation used, it is estimated that the second plate in parallel is only about 10% effective. The total installed damping is thus predicted to be on the order of 16% of critical--subject to the approximations noted.

3.3 Dynamic Similarity

The use of a scale model for the investigation of vessel motion control presents a number of difficulties even though testing in waves is conventional in seaworthiness research. It was desireable to plan tests with a wave height as large as possible to make the damping effects most visible. Then, with realistic values of wave length:height ratio, the wave is longer with respect to the craft dimensions than would be the case at full scale. The tests were planned to be safely within the capability envelope of the wave making facility. However, it was appreciated that dynamic similarity of the tests with lighters in the range of 150 ft. long and greater would require the steepest possible waves.

An example at this point will illustrate--with further discussion to be supplied after the test results are presented. If the full scale lighter is about the size of an LCU type it will have a nature period in heave of about 3.0 seconds (f=0.33 hertz). The wave length, assuming sine waves, to excite a resonance condition will be λ equal 44 ft. While such waves would have a fairly low height, they would occur frequently, and the amplitudes are not altogether insignificant. This wave length is but a fraction of the length of the lighter--approximately 0.33.

By contrast, the model with a predicted heave natural period of 0.74 seconds is excited at resonance by a wave of 2.8 ft. For a reasonably proportioned wave the wave height would be only about 1.5 inches. Since it was expected that visual observation of the effects of the damping appendage would be useful, extremely small waves were not programmed.

3.4 Sequence of Testing

Instrumentation was installed providing for wave and model measurements. A wave measuring probe was mounted on the carriage just ahead and to the left of the model. This location makes phase relationship determination from the data difficult but was a time saver during test setup. The carriage contained a potentiometer-type pickup for model displacements. The output of both instruments was recorded on strip charts.

The mounting of the model was intended to permit only the single degree of freedom in heave as waves were encountered. Other motions were restrained by locking out the additional articulations available on the test carriage. This test aim was not completely achieved due to flexibility in the supporting post and in the plate at which the post engaged the hull.

The first set of runs in waves was made with dampers installed. The range of wave lengths from 8-16 feet was covered with several wave heights, nominally three and six inches. The wave generating facility produced a lesser amplitude in all cases. In addition to exploring this range of wave conditions, two values of model displacement were tested. Actual draft measurements were made, with two and three inches freeboard being the reference condition, corresponding to 237.2 and 171.2 pounds respectively. Actually for the model with dampers installed, greater weight was required to reproduce the reference displacement conditions due to the previously noted contribution of the slats to buoyant forces. The latter values were 350 and 285 pounds.

Additional runs covered the bottom row of slats only of the damper at the light displacement condition and the bare hull at both displacement conditions. In the bare hull tests the range of waves lengths was expanded to six feet and a number of intermediate points on the spectrum of wave length were added as available time in the model basin permitted. Altogether 27 runs were made.

Following a preliminary analysis of the test data, during which the contribution of the dampers to the observed motions was unclear, a measurement of damping in heave was found to be in order. The model support was the same as

in the wave tests and motions were similarly recorded. Measurement of damping was made by the declining amplitude method. A pumping force was applied by hand to build up the oscillatory amplitude to about one inch. Following the removal of pumping action a free vibration enabled the measurement of the logarithmic decrement and the natural period of heave motion. All configurations of damping device and displacement were covered.

Section 4

DECLINING AMPLITUDE RESULTS

This test provides essential information for interpreting the response to wave excitation although it obviously does not lead to direct conclusions on the merits of a heave damper. The properties in question, namely damping and natural frequency in heave (which includes added mass), can be readily measured with sufficient accuracy to make the discriminations consistent with the scope of this effort.

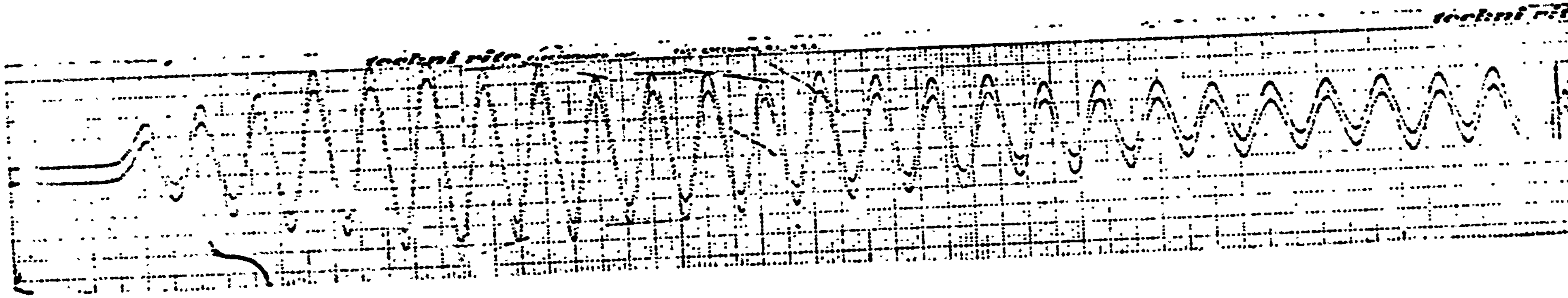
4.1 Natural Frequency Measurement

Typical recordings of the instrument output during these tests are shown in Figure 4 . In all cases the first few cycles show an increase in amplitude as the pumping force is applied. The point at which release takes place and free vibrations ensue is visible in all tests.

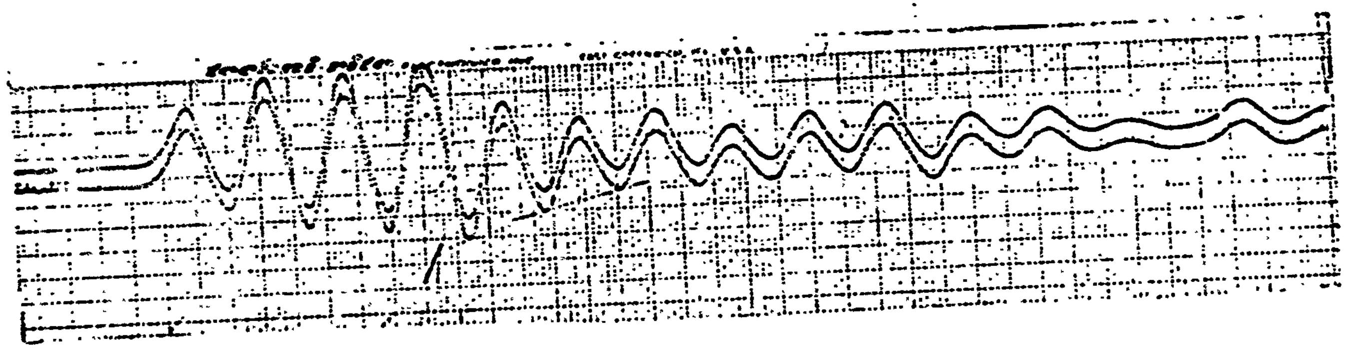
Consistency of the measured period for comparable runs was very good. The spread was on the order of one-half percent, which is about the best precision that can be obtained from the graphical records. The results are as follows:

Configuration	Period-sec.	Freq.-hertz
1. Bare hull-light	1.05	0.955
2. Bare hull-heavy	1.055	0.95
3. w/damper, single row	1.20	0.835
4. w/damper, double row	1.30	0.77

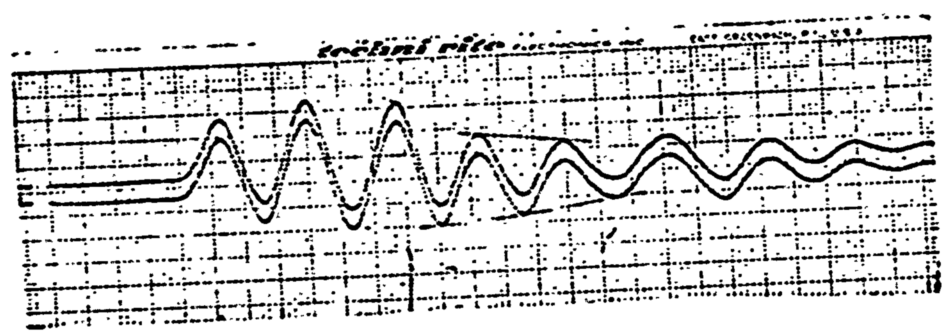
In the case of the configurations with dampers, the difference in period between the light and heavy displacements was hardly perceptible. For the bare hull the difference is nearly so. In the former case the differential between light and heavy conditions, being an arbitrary amount in any case, is a much lower fraction of the total mass.



a – Bare Hull



b – Hull with Lower Slats Only



c – Hull with Two Slat Rows

FIGURE 4 DECLINING AMPLITUDE TEST DATA

It can be noted that the measured natural frequency for the case of the bare hull is lower than the predicted. The discrepancy in the prediction contributed to the omission of a wave test at the point of resonance. Obviously the initial value of added mass explains much of differential between predicted and measured frequency.

4.2 Added Mass Determination

The measured frequencies provide a convenient determination of added mass in heave. It should be noted, however, that the added mass coefficients reported here are applicable to the frequency at which the natural heave occured and this property varies with frequency. For the bare hull, as ballasted, the full-scale area of the waterplane is 406 sq. ft. which scales down 11.3 sq. ft. for the model. The corresponding value of apparent mass is 650 pounds from which the actual value of 237 pounds can be subtracted to obtain 413 pounds of added mass. This is a ratio of heave added mass to displaced mass of 1.74--which conforms extremely closely to the data shown in the NCEL analysis of heave motions (Figure 3 of Reference 1).

In the case of the hull with damping appendage the added mass is 650 pounds, or 1.86 for the ratio of added to displaced mass. This particular value should be adjusted to account for the anomoly in the test article--namely that the buoyancy of the plywood slats resulted in a far heavier vessel for the case with damper installed as compared to the bare hull. The added mass of 650 pounds, which includes the additional water in motion due to the slats, represents a ratio of 2.7 when referenced to the bare hull.

4.3 Measured Damping By Logarithmic Decrement

The measurement of motion amplitude during free heaving motion can be converted to damping values in a direct way. The decrease of amplitude during damped free vibrations from one cycle to the next follows the law of geometric progression:

$$\frac{z_{n+1}}{z_n} = \exp\left(\frac{-2\pi\frac{C}{C_{cr}}}{\sqrt{1-\left(\frac{C}{C_{cr}}\right)^2}}\right)$$

Then the difference between the natural logarithms of two successive amplitudes is:

$$\frac{2\pi \frac{C}{C_{cr}}}{\sqrt{1 - \left(\frac{C}{C_{cr}}\right)^2}}$$

which reduces to:

$$2\pi \frac{C}{C_{cr}}$$

when the damping is small, i.e. less than about 0.10 of critical.

The maximum number of measurements of amplitude on the recorded motion pickups was used to offset the disturbances in the damped sine waves. The disturbances were imparted to the water surface during pumping prior to the free vibration but were essentially averaged out during multiple runs. The damping values as a fraction of critical damping are found to be:

Model	C/C_{cr}
1. Bare Hull	0.013
2. w/Damper, single row	0.051
3. w/Damper, double row	0.080

Section 5

PERFORMANCE IN WAVES

The plan of testing was designed to cover a useful range of wave lengths and heights for the hull only and with damping appendage. Variations in damper were included along with displacement variations. The heavier conditions were intended to bring the model closer to a condition of resonance with the wave excitation. The desire to use the maximum wave height for best visual perception of the damping effects and the limits on wave steepness at the model basin had the effect excluding some tests at shorter wave lengths. In retrospect it appears that tests at wave lengths as short as half the hull length would have been illuminating.

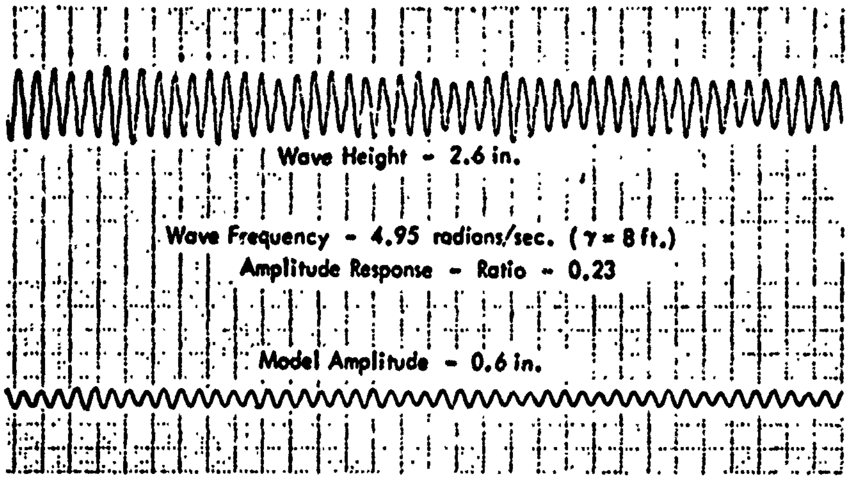
5.1 Response Variation With Wave Length

With the model mounted on the stationary carriage, and the wave making facility producing steady conditions, amplitude measurements were taken. Typical records are shown in Figure 5 . The degree of pitch motion present and coupled to heave is an unknown quantity, not having been measured. It can be said that the restraints against all motions other than heave were not completely effective due to elasticity in several support members. Nevertheless, there does not appear to have been sufficient pitch to alter the fundamental heave phenomena.

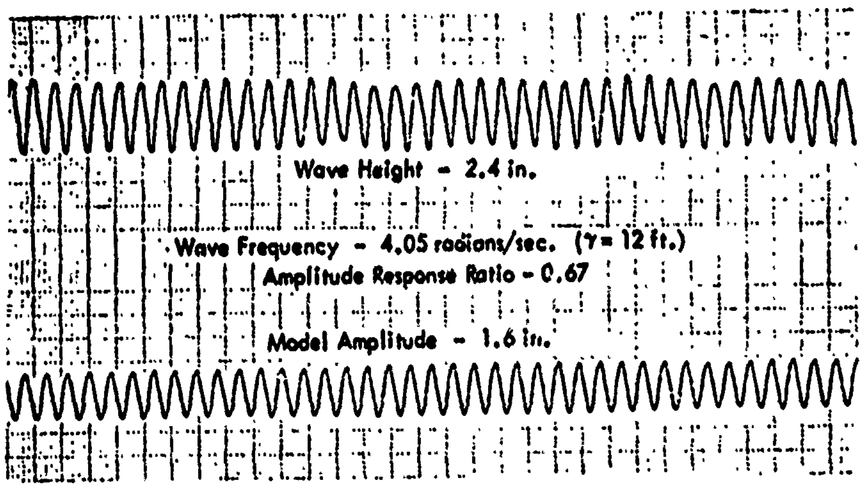
The results for all tests are plotted in the usual form of a response amplitude operator in Figure 6. This is the non-dimensional ratio of model heave amplitude to wave amplitude. The horizontal axis is wave circular frequency in radians per second. Each wave frequency has an associated wave length ( ) related to frequency according to:

$$\omega^2 = \frac{2\pi g}{\lambda}$$

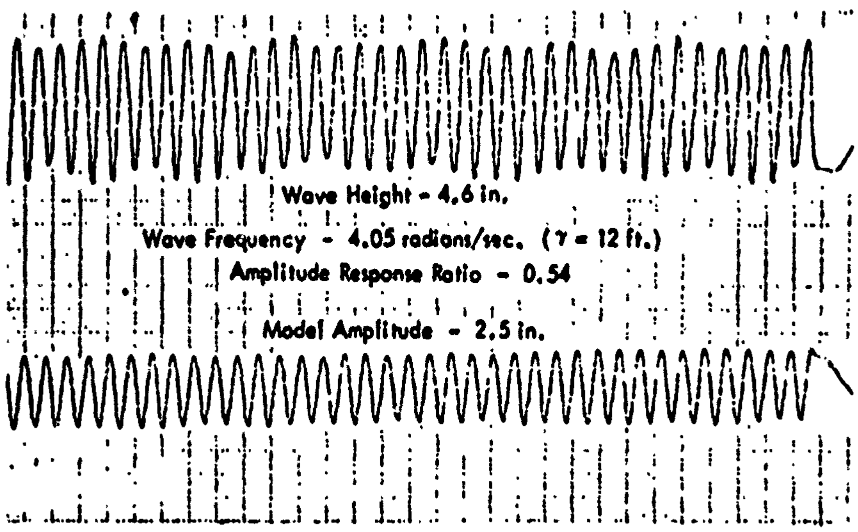
The ratio of wave length to hull length is also indicated on the horizontal axis since this mode of data presentation is also widely used.



a. Hull With Double-Row Damping - Light Displacement



b. Hull With Double-Row Damping - Heavy Displacement



c. Bare Hull - Light Displacement

Figure 5 Typical Test Data From Wave Runs

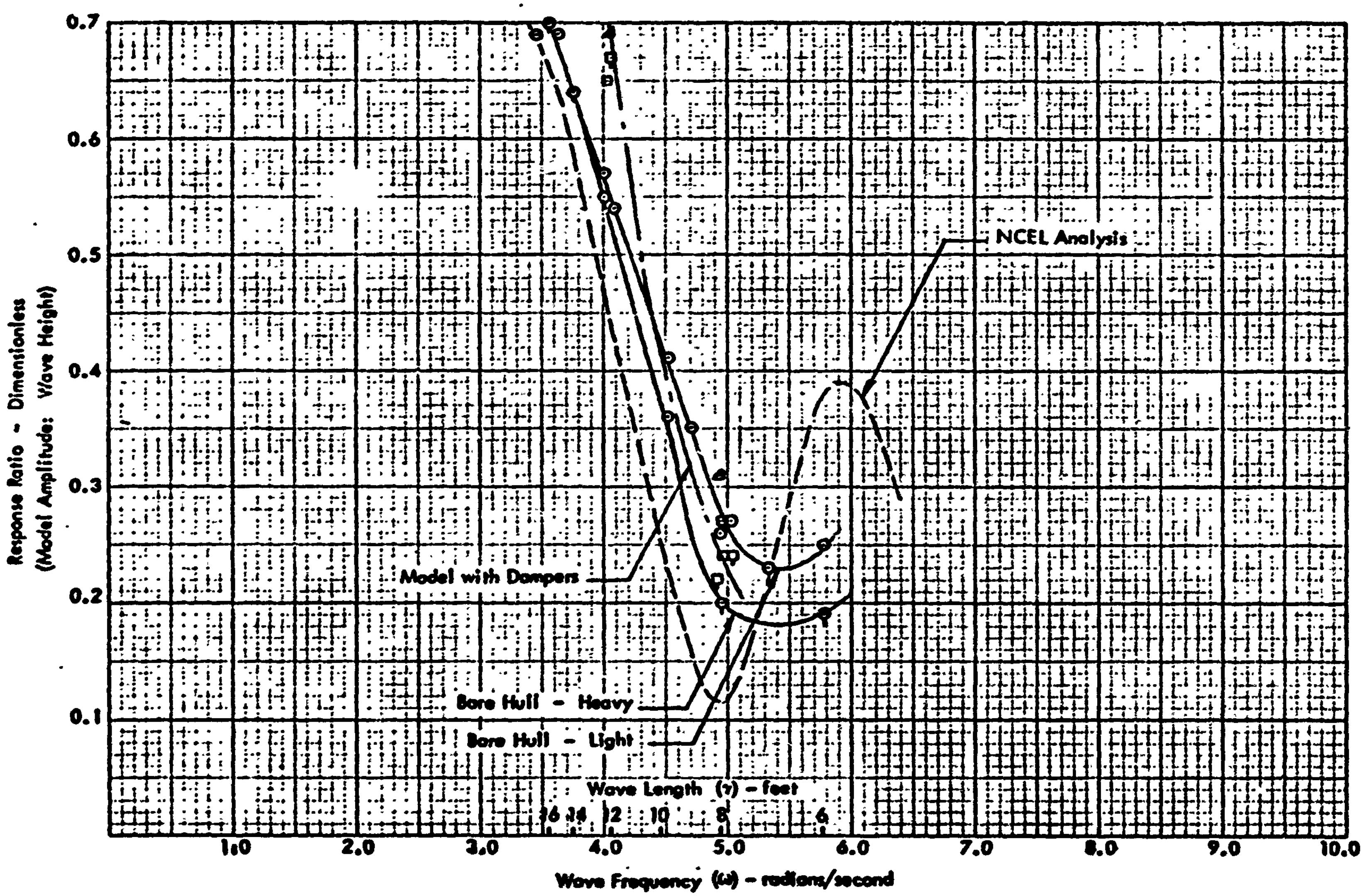


Figure 6 Response Amplitude Operator

The general form of the results is a decreasing value of the heave response amplitude with increasing values of wave frequency--out to about five radians per second. The case of the bare hull at light displacement, for which sufficient test points are available, produces a rather well defined curve. Between 5.0 and 5.8 radians/sec. a reversing trend may be seen to emerge. The curve for the bare hull at heavy displacement does not have as many test points, but again a reversing trend is apparent.

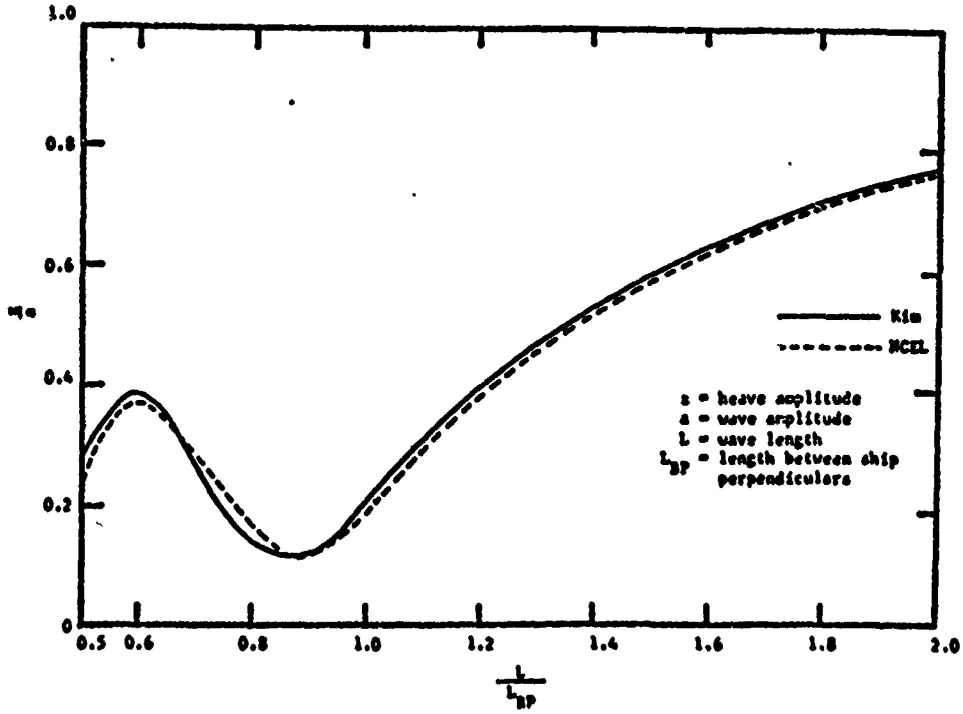
The heave response for the model with damper installed takes the same general form out to about 5.0 radians/sec., beyond which, unfortunately, no test points were taken. It is here in the frequency region where the usual form of the response amplitude operator curve exhibits a peaking that the benefit of damping should be realized. Further discussion on this matter follows in the next section. However, before leaving Figure 6 it may be noted that the model with damper shows greater heave response at the test points around [illegible].1 radians/sec. The ratio of λ:L is fairly high here being about 1.6. In this region of the curve the response should be governed by the magnitude of the excitation of the waves without appreciable influence of proximity to resonance conditions. A possible explanation is that the damping slats, being in close proximity to the hull, were acted on by orbiting water to augment the forces exciting heave on the hull. The single row damper model is closer to the bare hull disposition of test points.

5.2 The Form of the Response Amplitude Operator Curve

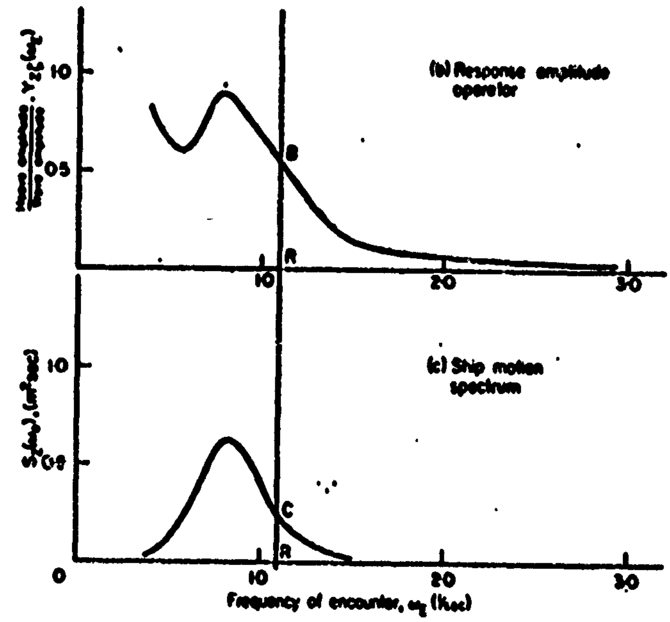
An assessment of the performance of the damping device in waves must focus on the response region beyond 5.0 radians/sec. The curve for the bare hull appears to have bottomed out and reversed the downward tendency. The damping device was unexplored in this region. There are some experimental and analytical indicators of what would have been encountered.

Analysis of uncoupled heave is covered in the NCEL analysis previously cited as Reference 1. The response curve obtained for heave motions contains a peak beyond the initial descent. This curve along with additional representative cases from Reference 2 are contained in Figure 7. Additionally the form of the curve can be substantiated by test results from Korvin-Kroukovsky's paper on pitching and heaving in waves, a reproduction of which is contained in Figure 8. This particular model was selected because it offered four points on the scale of λ:L, less data being shown for the other models.

The various referenced studies are directed to topics in seakeeping and do not regard the form of the heave response curve as a matter of special interest.

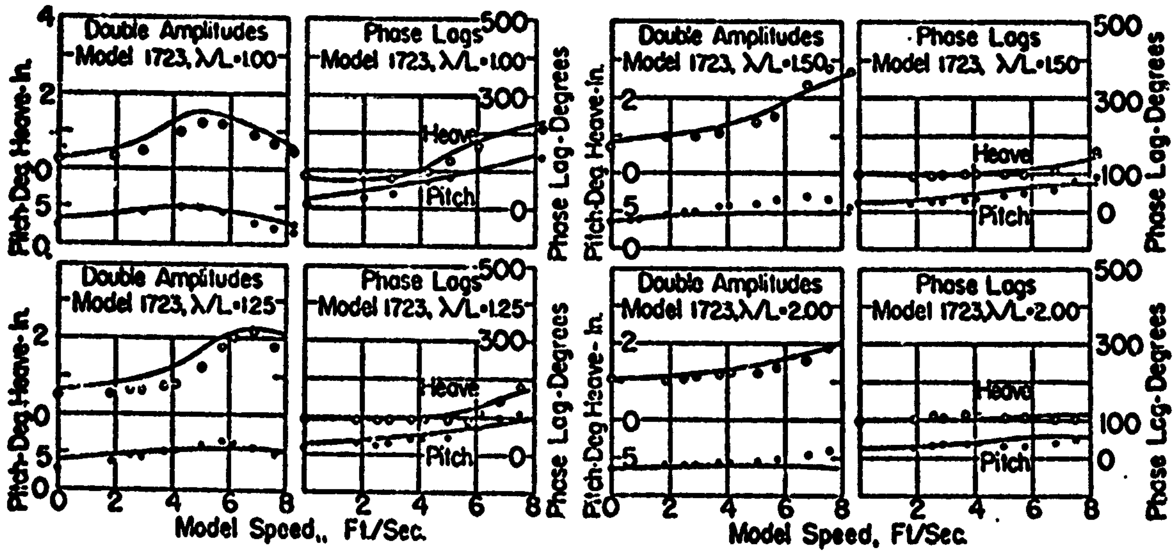


a. Heave Amplitude Ratio for Model Ship
(Reproduced from Reference 1)



b. Response Amplitude Operator from Rawson & Tupper
(Reproduced from Reference 2)

Figure 7 : Heave Motion Results



Motions of 5.7-Ft BTT Model 1723 (Destroyer) in Waves of 1.0, 1.25, 1.5, and 2.0 Model Lengths by 1.43 In. High

Figure 8 - Heave Results From Tests of a Destroyer Type Model (Reproduced from Reference 3)

However, in the present case an explanation of the form of the curve, and in particular the peak, is of critical importance. The formulation of equations of motion contain terms corresponding to the inertial, damping, restoring, and exciting forces as in simpler cases of single degree of freedom systems. It is apparent from examining the studies that two primary influences carry through to the heave response:

- The ratio of λ:L governs the exciting force in heave resulting in the maximum force for long waves and reaching a minimum when the ratio is near to one (or multiples thereof).

- The ratio of $\omega_e : \omega_n$ (the tuning factor of exciting frequency to natural frequency) will produce a peaking of the response in the vicinity of a value of one.

The length ratio effect is readily confirmed. Visual inspection of the NCEL curve shows a valley at about 0.9. This should fall at some value just short of one because the hull is not completely block-like. With a fine bow region an effective ship length should be less than the actual dimension and produce the equivalent of a unity ratio when the wave length is correspondingly less than the measured ship

length. Referring to the model test data, and analysis results, of the K-K work at zero model speed it may be seen that the response at λ:L = 1 is the least at about 0.15 for the amplitude ratio. This is quite close to the amplitude ratio of the NCEL model analysis.

The curve reproduced from Rawson and Tupper has a somewhat different form in that it does not go nearly as low and its peak following the depression is much higher as compared to the model data. The ship is not defined except in a passing remark to be 175 meters long. A naval auxiliary of known characteristics and 565 ft. long was assumed to be close to the representative ship. Taking account of the length reduction at the waterplane and compensating for effective length, the wave length should be close to 520 ft. and have a circular frequency of 0.62 radians/sec. Referring back to Figure 7 it may be seen that the depression in the curve does in fact occur close to this point along the horizontal axis. The natural period in heave for the ship is found to be 7.9 seconds for a circular frequency 0.80 radians/sec. Again, inspection of the curve substantiates the expectation and the peak is located at the predicted point in frequency.

Further consideration of the peak of the NCEL data confirms that it occurs at a tuning factor close to one. The natural period in heave for the 347 pound displacement with added mass and a water plane coefficient of 0.67 is found to be 1.15 seconds, for a frequency of 5.45 radians/sec. The wave length to excite resonance is 6.80 ft. for a ratio of λ = L of 0.69 which may be seen on the curve to be the approximate location of the peak.

Returning to the Korvin-Kroukovski data, resonance effects can be detected. In this case there are insufficient points at zero speed to construct a frequency response to include resonance. However, two heave amplitude response curves do exhibit a peaking, for λ : L = 1.00 and λ : L = 1.25. In the first case the frequency of encounter can be found at 1.80 cycles per second due to the wave speed of 5.4 fps and the model speed of 4.8 fps at maximum heave response. The encounter period is 0.56 seconds, quite close to the reported natural period of the model at 0.61 seconds. In the next longer wave the peaking is at about 6.5 fps and the frequency of encounter is 1.76 cycles per second. The encounter period is 0.57 seconds. This peaking of the heave response, even though coupled with pitch motions, at periods which are so close to the computed natural period in heave are taken as a point of evidence that resonance effects do produce a visible effect on forced heave motion.

A study of forces on a heaving ship at DTMB (Reference 4) has some bearing on the question of the form of a heave response amplitude operator curve.

Tests were performed by driving a hull in heave with an oscillator--in still water--and measuring the force to produce unit amplitude. The zero speed case is reproduced from the reference in Figure 9 . The published data on the 11.3 ft. model led to a natural frequency of 5.5 radians/sec. using an added mass factor comparable to the other models. An obvious resonance response appears on the curve although inverted when expressed in force terms. The explanation for the higher frequency of resonance can be had in the added mass term. The model weight was held to a minimum weight to keep the load on force measuring instrumentation low and the test displacement of 317 pounds is not a realistic simulation of full scale. However, the water volume in motion under the influence of hull form and size is not effected, thereby driving the ratio up. In any case, the force reduction beyond 5 or 6 radians/sec. must contribute to the peaking observed on wave-excited hulls in this region. The reference also contains phase measurements, which show the characteristic change at resonance, and damping coefficient variation with frequency.

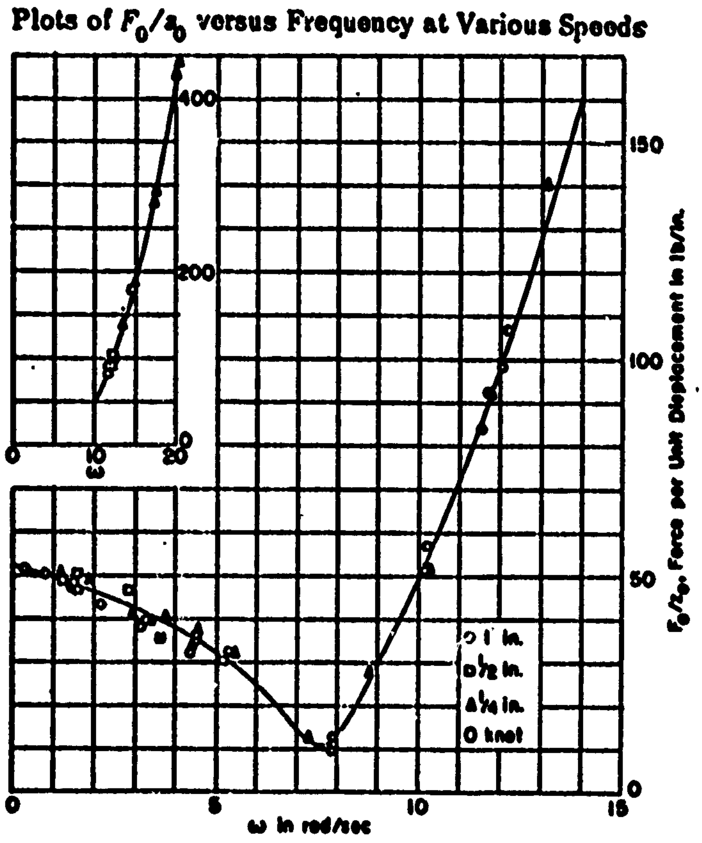


Figure 9 - Illustration of Reduced Force To Drive Heave Motion at Resonance
(Reproduced from Reference 4)

5.3 Resonance Influence on Wave Response

In presenting the test results on the basis of the ratio of wave length to ship length or on the frequency of wave excitation, any gross dissimilarity in natural frequency is not immediately apparent. Accordingly, the test results are replotted in Figure 10, this time with the tuning factor (the ratio of exciting frequency to natural frequency) as the horizontal axis. Measured frequencies from the declining amplitude tests were used. The form of the curves is not disturbed but the model with damper is shifted to the right.

The significant finding in the replot is the indication of the test points for the model with damper. The curve descends under the influence of the wave/hull ratio. Several test points beyond $\omega_c/\omega_n = 1$ have low values comparable to the bare hull test points. The reversing trend toward a peak response has not developed. While undoubtedly there will be some level of peaking in the curve as the depression is passed, the amplitude ratio should not rise as high as in the other cases. The consequences of damping, which as a general rule are evident in the resonance region, appear to be a suppression of the peaking.

In addition to some damping of the peak, another effect is present. A reasonably constant relationship has been found for models of various hull forms between the frequency at which wave length equals ship length and the frequency at which resonance occurs. The ratio is approximately 0.8. In the case of the present tests the ballast level was higher than the design condition and the ratio of the two frequencies for the bare hull was 0.87. The two frequency values are at the depression and peak respectively. However, in the case of the model with damping appendage, the ratio does not follow the pattern. The resonant frequency of of the model was lowered by about 20%, thus nearly superimposing the two important frequencies.

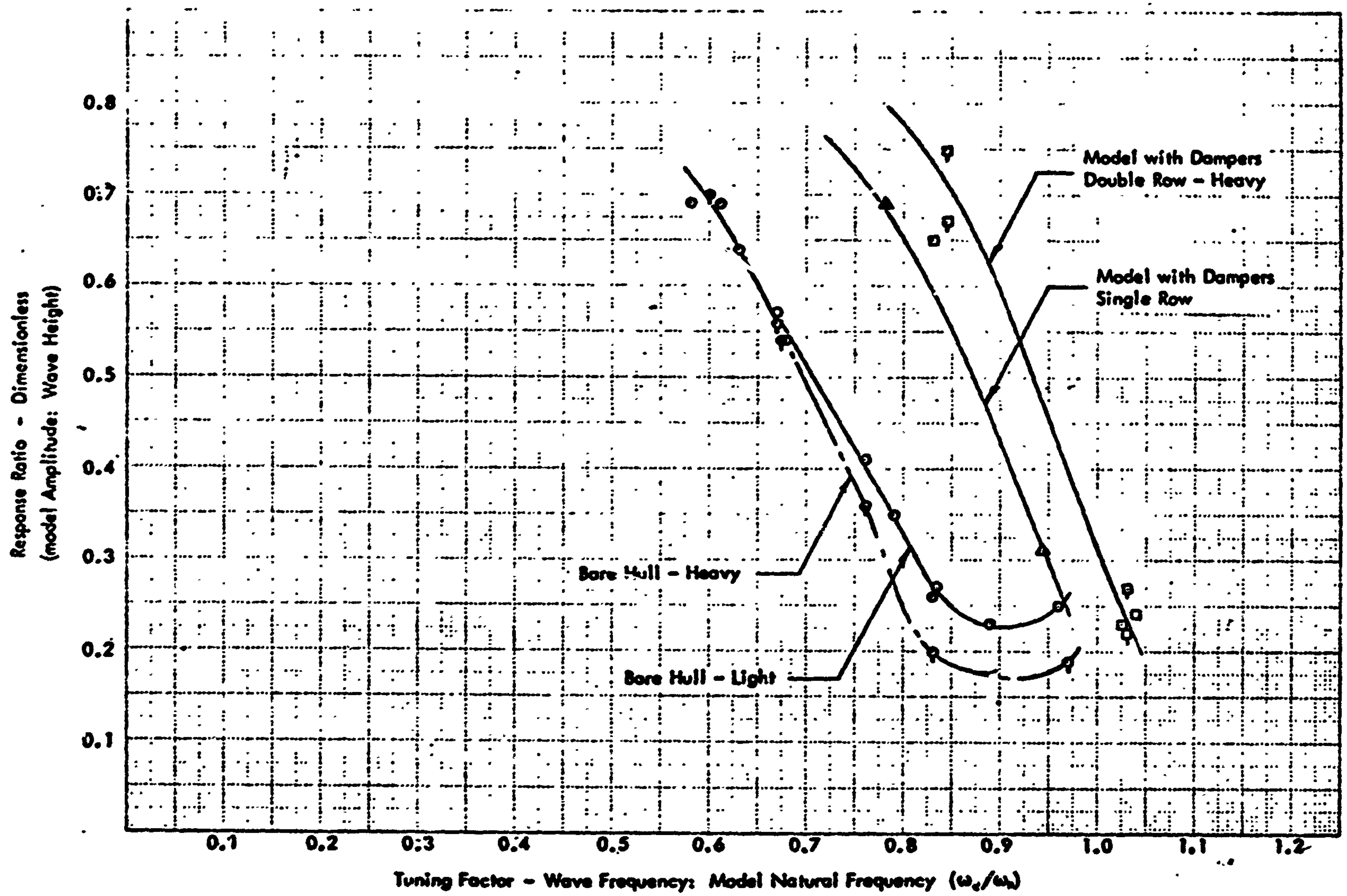


Figure 10 Influence Of Resonance On Model Heave Response

Section 6

FULL SCALE IMPLICATIONS

It is necessary to consider the situation at full scale before attempting to firm up conclusions on the ultimate utility of the concept of damping augmentation for a ship to shore lighter. It was evident in the tests that no performance gains were realized at the longer wave conditions, say from 1.3 to 2.0 (and no doubt upward) of the model length. Unquestionably the amount of heave exciting force--for the model geometry and mass--is related to the wave length ratio. Resonance is far enough removed from this range of conditions that response curves here are not effected by damping. Focusing on the peak of the heave response amplitude operator curve, the case for existence of a damping influence appears strong. The probable importance of the peak will be considered.

6.1 The Wave Length Ratio at Full Scale

A lighter about the size of an LCU is a suitable starting point. Smaller craft are most appropriate for early waves of an amphibious operation to gain dispersion in moving against hostile shores. However, productivity in moving cargo goes up markedly as size increases and the LCU has played a vital role in the logistical phase of ship to shore operations. Consider a length of 140 ft. as representative for existing classes and any foreseeable replacement class.

Data on a sea spectrum are conveniently available in the NCEL work cited as Reference 1. A six-foot wave height marks the entry into Sea State 4 and is an obstacle to productive cargo flows. Spectrum data indicate an average wave length of 78.8 ft. for this wave. Lower wave heights, and correspondingly shorter wave lengths, can be expected to occur for a larger fraction of the time but as the wave height diminishes the obstacle to cargo transfer also lessens. The six-foot wave could very well be a satisfactory design point.

The wave length to ship length ratio for the wave and hull in question is about 0.55. For the 1610-Class of LCU the ratio is 0.58. It is clear that the wave length ratio of interest at full scale is in the vicinity of the peak in the typical response amplitude operator curve.

6.2 Resonance Frequency at Full Scale

The position of the peak on the response curve should be checked for full scale. The 1610-Class of LCU, using the established added mass ratio of 1.8, has a resonant frequency in heave about 2.1 radians/sec. Using the relations for sine waves this is a wave 44 ft. long. This would indicate that the peak at full scale is somewhat further separated from the depression of the response curve than for the case of the scale models. This being the case a rather large band of wave lengths of importance (i.e. probable frequency of occurrence) fall beyond the depression in the response curve and the role of damping is significant at full scale. Stated in corollary terms, the cases of long wave length with respect to ship length are of little significance at full scale and will occur most infrequently. The cases where damping has little of no effect are thus not near the design point.

6.3 Light Vessel Condition

Up to this point the mass of the heaving body has been the full load displacement. Obviously a lighter spends much time at a light displacement condition. This raises the natural frequency. Concurrently a shorter wave will produce the resonant condition, say a reduction from 44 to 37 ft. This is not in proportion to the difference in displacements since the added mass should not be materially effected. The consequence is that the peak in the response curve is further removed from the depression. Similarly the six-foot wave will be displaced further off the resonance peak and toward the depression. However, the peak response will be excited more frequently even though at a lower level of excitation.

REFERENCES

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3 B. V. Korvin-Kroukovsky et al., Pitching and Heaving Motions of a Ship in Regular Waves

4 Paul Golovato., A Study of the Forces and Moments on a Heaving Surface Ship., Hydromechanics Laboratory Research and Development Report., Sept. 1957., Report 1074.